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Air cooling af a horizontally grooved turbine blade model with covering metal sleeve.

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AIR COOLING OF A HORIZONTALLY GROOVED TURBINE BLADE MODEL WITH COVERING METAL SLEEVE

Submitted to the Graduate Faculty
of the
University of Minnesota

by
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Lt. U.S.N.

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SUMMARY

A static test on a particular air-cooled turbine blade model was conducted at the University of Minnesota in July, 1951. The blade model utilized cooling air which was ducted into the blade near the leading edge, thence into horizontal, or chordwise, grooves between the blade and a thin metal sleeve attached to lands on the blade. Cooling air was discharged at the trailing edge of the blade, where an opening was provided in the sleeve.

Mach numbers in the flow around the test blade were from .4 to .5 with tests being made at gas temperatures at about 800° F., 1000° F., 1200° F., and 1420° F.

The following conclusions were reached:

1. At gas temperatures of about 1420° F., a temperature reduction of 630° F. was experienced near the trailing edge, and a reduction of 890° F. was found near the leading edge, for a cooling air flow rate comparable to 1.67% of combustion air.

2. The blade configuration tested possessed excellent cooling characteristics and showed an economy of

cooling air use compared to available data on other cooling configurations.

3. Greater temperature reductions were found at high gas temperatures than at low gas temperatures, with constant rate of cooling air flow. The rate of increase of temperature reduction with gas temperature increase appeared to be linear over the range tested.

INTRODUCTION

The broad problem in the field of gas turbine operation, with respect to turbine blades, is that of developing a blade capable of withstanding high stresses in a region of high temperatures. Since there are today many hundreds of turbines operating, it is evident that some success has been met in this development.

There is very little which can be done to reduce the stresses associated with the contrifugal forces of the high speed turbine. It is also highly desirable to operate these turbines at the highest permissible limits of temperature. Therefore, cooling of the turbine blades by some outside means has been under considerable investigation recently, as a method of permitting higher turbine gas temperatures. Some of the advantages which may accrue from effective blade cooling are increased power, prolonged blade life, and use of less critical and expensive materials in blade construction.

This report describes the static test of a turbine blade model which was designed to give high economy of cooling air by using the air as a protective layer between the blade body and a covering metal sleeve.

I DESCRIPTION OF TEST BLADE AND EQUIPMENT

Fig. (1) shows a sketch of the turbine blade model, and Fig. (14) shows a photograph of the blade with the covering metal sleeve attached. The blade was machined from mild steel. No attempt was made to give a twist to the blade, and for simplicity of lathe machining, the air-foil surface was formed of two circular arcs filleted as seen in Fig. (1). The grooves are .025 inches deep; the sleeve is .033 inch rolled black iron sheet. The materials used were chosen because of their ready availability and machinability. The sleeve was formed around the blade, and attached with counter-sunk rivets and screws, which were ground off to be flush with the surface. Total surface area of the blade was 33.8 sq. in. Blade height was 4½ inches.

Eleven holes for iron - constantan thermocouples were drilled about one-third of the depth of the blade. Only seven of these positions were employed in the tests.

Great care was exercised in drilling the small one-sixteenth inch holes from the leading edge to the main

cooling air duct, for misalignment of those holes could cause maldistribution of cooling air to the grooves on each surface.

The test section contained two uncooled blades similar to the test blade, and is shown schematically in Fig. (2). A photograph of this section is given in Fig. (13). The test blade was mounted on a pedestal arrangement to allow its easy insertion into the test section between the two uncooled blades. The blades, with the surfaces of the test section, formed a cascade, making the flow turn an angle of about sixty-four degrees. Each uncooled blade had a thermocouple installed near its leading edge.

The tests were run in an especially designed Gas Turbine Test Cell in the Mechanical Engineering building of the University of Minnesota. The photograph of Fig. (12) shows the control panel, and Fig. (11) shows the test cell interior. There was a Lycoming Model O-435-T air cooled engine, rated at 162 HP at 2800 RPM driving an air compressor, which was a 7.48 : 1 gear ratio supercharger from an Allison V-1710 aircraft engine. The air delivered by the supercharger to the large manifold was ducted to the com-

bustion chamber of a single Allison J33-A-17 turbojet engine burner. Combustion was started by a spark-ignited acetylene flame, and combustion temperatures were controlled by the burner fuel pump bypass, for regulating fuel flow, and by the Lycoming engine throttle, which determined engine RPM, hence supercharger flow rate. Number one diesel fuel was used in the combustion chamber.

The test blade was located in the test section about eleven and one-half inches downstream of the combustion chamber exit.

All thermocouples used were iron - constantan, and were read on a Brown Recording Potentiometer having a scale from 0 - 1600° F.

Cooling air was supplied from the compressed air system of the Mechanical Engineering building. Pumping capacity of the system was greater than the maximum flow rate used, and the supply was available at all times between 80 and 100 psig. Cooling air flow rate was determined from a Fischer and Porter "Flowrator" with a tube size # 5A-25. Cooling air initial temperature was measured by thermocouple.

Air flow to the burner was measured at an orifice on the intake side of the engine-driven compressor. The orifice was 5.6 inches diameter, in a circular duct eight inches in diameter. Static pressure taps were installed one diameter upstream and one-half diameter downstream of the orifice.

Fuel flow to the burner was measured on a fuel "Flowrator" tube # 5A-60, mounted on the control panel.

Temperature and pressure were measured in the test section four and one-half inches upstream of the test blades. A total pressure tube and a static pressure tap were employed, and a shielded total temperature probe housed an iron - constantan thermocouple. This temperature probe read consistently lower than the uncooled blades of the test section, however, so it was considered of value only as a "reference" temperature. At a constant burner air flow, any desired temperature could be obtained and held constant with ± 5° F. on this "reference" probe by controlling the burner fuel flow.

II TEST PROCEDURE

Test procedure was simple. Reference temperatures of 800° F., 1000° F., 1200° F., and 1420° F. were successively obtained on the shielded temperature probe. At each reference temperature the flow of cooling air was varied, and readings were taken of all instrumentation as shown in Table I. Great care was exercized in order that equilibrium be reached with each new rate of cooling air flow before readings were taken. A curve is shown in Fig. (7) for a temperature-time check on thermocouple #5 at reference temperature of 1420° F., and the final point of this curve agrees with the reading taken at the beginning of that series of runs, showing that the procedure used was satisfactory.

III DISCUSSION OF RESULTS

(a) Results of the present investigation

The data are tabulated in Table I. Figs. (3), (4), (5), and (6) show plots of the recorded temperatures of all thermocouples on the test blade vs the weight rate of cooling air flow as determined from the "Flowrator", and represent graphically the results of the tests. It may be noted from the figures that at each reference temperature there was a marked blade temperature reduction for each thermocouple location. No thermocouples were located forward of the main cooling air duct because of space limitations. Thermocouples #1 and #2 consistently read very nearly the same temperature; a natural result since they were both near the duct of incoming cooling air. Temperatures of the points on the concave side of the blade (even numbered points) read slightly lower than those on the convex side, possibly because of greater resistance to flow in the longer grooves of the convex side, which may have caused less cooling air to flow in those grooves.

The distribution of temperature along the blade finds the hottest part at the trailing edge, the coolest

part at the incoming air duct near the leading edge, with a maximum temperature difference between hot and cool points of 300° F. The temperature of the cooling air rose as it was heated in its passage along the grooves.

Thermocouple #5 was chosen as a representative point for comparison of temperature reductions at different air flows and uncooled temperatures, for it represents a point removed from the great cooling near the leading edge, and is near the hotter point of the trailing edge. Fig. (8) shows a plot of temperature reduction vs cooling air flow for this thermocouple at various reference temperatures of the hot gases. It was found that temperature reduction increased with flow rate of cooling air, but that after a point, the rate of this increase was small.

An interesting cross-plot of Fig. (8) is shown in Fig. (9) as a set of curves of temperature reduction vs uncooled temperature, for the various rates of cooling air flow. This cross-plot shows that for the region of the tests the temperature reduction at a given weight rate of cooling air flow increased almost linearly with the uncooled temperature. If this linearity holds into regions of higher temperatures, a very rewarding employment of cooling air

might be experienced in the neighborhood of 2000° F. and over.

Fig. (10) shows the temperature distribution along the blade at 1420° F. reference temperature, with various rates of cooling air flow. This figure pictures a trend already mentioned - increasing temperatures toward the trailing edge as the cooling air is heated up. The close agreement of the temperatures along the two surfaces is an indication that no major distribution errors in the cooling air flow occured between the two grooves.

While no data were taken to permit calculation of the sleeve temperature, it was not considered that the sleeve will be a critical part of the blade with regard to temperature, because the amount of blade cooling present makes it obvious that a sizeable heat transfer is going on between the hot gasses of combustion and the sleeve; for this condition to occur, there must be a large temperature gradient between these hot gases and the sleeve. Furthermore, in a turbine, the sleeve as constructed would not have to carry centrifugal stress loads as high as the blade body because of its several support lines furnished by the lands.

(b) Comparison with other investigations

Since the blade model tested was large compared to turbine blades normally used in aircraft engines, a method for comparing the cooling required was considered in order to evaluate the results in terms of other investigations concerned with air-cooled turbine blades for aircraft. The heat flow equation $Q = hA\Delta T$ was used for this purpose, and the blade size used for comparison was the J33 turbine blade, having an area of about 14.8 sq. in. Test Blade area was 33.8 sq. in.

In the heat flow equation, the variables to be considered were the film heat transfer coefficient, "h", from the hot gases to the sleeve, and the blade area, A. The same ΔT was considered for both sizes of blade, and the ratio of heat flows to each blade was estimated. It was assumed that the rate of cooling air flow required would be proportional to the rate of heat flow to the blade sleeve.

For the test blade: $Q_1 = h_1A_1\Delta T$

For blade of 13.8 sq. in.: $Q_2 = h_2A_2\Delta T$

and $Q_1/Q_2 = (h_1/h_2) \times (A_1/A_2)$

From Ref. (f), page 106, a relation for the film heat transfer coefficient, h, is given for plane surfaces, and was assumed to hold approximately for the sleeve surface:

$h = .055\ (k/L)\ (N)^{.75}$, where

k = heat transfer coefficient of the gas
L = representative lenght
N = Reynold's number

Substituting the relation for "h" into the expression for heat flow ratio,

$$Q_1/Q_2 = (A_1/A_2)\ (L_2/L_1)\ (L_1L_2)^{.75}$$

A heat flow comparison was made between the test blade and a geometrically similar blade to it, but which had the same area as the J33 blade:

$$Q_1/Q_2 = 2.05$$

It was then assumed that the larger test blade had required 2.05 times as much air for cooling as the smaller blade would have required. There was then a basis for a rough comparison of weight of cooling air to weight of combustion air.

On a J33 engine there are 14 burners of the type

employed in the tests, and there are 54 turbine blades having areas of about 14.8 sq. in. each. So that it was calculated if the 54 blades of the J33 were similar to the test blade, and air-cooled as the test blade; at an engine airflow fourteen times that of the tests, and a temperature of abour 1420° F. at the turbine inlet, the cooling conditions found in the test blade would be found in the smaller blades at cooling airflows of .487 those of the test blade.

Using the maximum flow rate of cooling air, 1.204 lb/min, which was employed in the test blade at 1420° F. reference temperature, it was seen that the smaller blades should have been using a total of .528 lb/sec of cooling air, and that the ratio of cooling air weight to combustion air weight would be 1.87%. The temperature reduction would have been the same as for the test blade, according to the preceeding calculations.

Care must be taken not to accept the above comparisons as having been proved by these tests. However, the comparisons do indicate that excellent results may be expected by use of the test blade cooling configuration on actual turbine blades.

Table II shows the results of several investigations on air cooling of turbine blade models. It is seen that the blade model of the present investigation shows excellent possibilities with regard to temperature reduction of blade, and weight ratio of cooling air flow to burner air flow.

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IV CONCLUSIONS

The following conclusions have been drawn from the tests conducted on the horizontally grooved air cooled turbine blade model with covering metal sleeve:

1. At combustion gas temperatures of about 1420° F., a temperature reduction of 630° F. was experienced near the trailing edge, and a reduction of 890° F. was found near the leading edge, for a cooling air flow rate comparable to 1.07% of combustion air.

2. The blade configuration tested possessed excellent cooling characteristics and showed an economy of cooling air use compared to data on other cooling configurations.

3. Greater temperature reductions were found at high gas temperatures than at low gas temperatures, with constant rate of cooling air flow. The rate of increase of temperature reduction with gas temperature increase appeared to be linear over the range tested.

TABLE I

TEST DATA AS RECORDED FOR AIR-COOLED TURBINE BLADE MODEL HAVING COVERING METAL SLEEVE. TEMPERATURES ARE IN DEGREES FARENHEIT AS DETERMINED FROM IRON-CONSTANTAN THERMOCOUPLES.

BAROMETER: 29.15 IN. HG. TEMP: 88 °F

COOLING AIR PRESSURE AT FLOWRATOR	COOLING AIR	TEST SECTION TOTAL PRESSURE	TEST SECTION STATIC PRESSURE	BURNER AIR INTAKE ORIFICE	BURNER FUEL	THERMOCOUPLES IN TEST BLADE							UNCOOLED BLADE	UNCOOLED BLADE	"TOTAL" OR REFERENCE TEMPERATURE	BURNER AIR ROOM INTAKE	COOLING AIR AT FLOWRATOR
(GAGE) LB/IN²	FLOWRATOR READING	P_o IN. HG.	P_s IN HG.	ΔP IN. H_2O	LB/HR	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}
60	8.05	32.4	.4	15.6	88	320	320	400	390	440	425	450	835	805	800	105	95
40	6.10	32.4	.4	15.6	88	370	370	450	440	500	480	510	840	810	810	105	95
20	4.65	32.3	.4	15.6	89	460	465	540	525	595	575	600	840	810	800	105	95
10	3.51	32.3	.4	15.6	88	520	525	595	575	640	630	650	840	810	800	105	100
2.5	1.82	32.3	.4	15.6	88	660	665	735	690	750	750	775	840	810	800	105	100
0	0	32.3	.4	15.6	87	825	825	825	800	820	830	840	840	800	800	105	100
60	7.8	32.8	.4	15.6	115	400	400	500	490	565	540	580	1035	1030	1000	100	90
40	6.1	32.8	.4	15.6	115	460	465	570	550	640	620	660	1035	1030	1000	100	90
20	4.65	32.9	.4	15.6	115	565	570	675	650	745	725	770	1035	1030	1000	105	90
10	3.45	32.9	.4	15.6	115	680	690	780	750	840	835	880	1040	1040	1000	105	95
2.5	1.82	32.9	.4	15.6	115	840	845	920	880	960	950	990	1045	1045	1000	105	100
0	0	32.9	.4	15.6	115	1020	1020	1020	995	1025	1030	1050	1050	1050	1000	105	100
60	7.7	33.5	.4	15.6	146	470	475	610	590	695	665	720	1220	1250	1200	105	85
40	6.0	33.5	.4	15.6	146	550	560	695	670	780	760	810	1225	1260	1200	105	90
20	4.55	33.5	.4	15.6	146	680	685	825	790	920	890	950	1230	1260	1200	105	90
10	3.38	33.5	.4	15.6	146	825	830	950	910	1035	1020	1070	1240	1275	1200	105	90
0	0	33.5	.4	15.6	146	1220	1220	1220	1190	1230	1230	1260	1250	1290	1200	105	90
60	7.6	34.0	.5	15.6	170	530	540	710	660	800	770	830	1395	1480	1410	105	90
40	5.95	34.0	.5	15.6	170	620	625	800	750	895	870	935	1400	1490	1415	105	90
20	4.35	34.0	.5	15.6	170	800/810	810/820	980/1000	930/945	1080/1100	1050/1070	1120/1130	1410	1500	1420	105	90
10	3.35	34.0	.5	15.6	170	970	980	1150	990	1240	1210	1270	1410	1490	1400	105	95
0	0	34.0	.5	15.6	170	1425	1420	1420	1410	1440	1430	1465	1420	1500	1410	105	95
60	7.6	34.0	.5	15.6	170					820			1400	1490	1420	105	95

TARE = .5 TARE: .15

SMALL TABLE IS TEMPERATURE-TIME CHECK ON THERMOCOUPLE # 5 AS BLADE WAS COOLED FROM 1440 °F TO 820 °F WITH COOLING AIR AT 60 PSIG

MIN	SEC	T_5	MIN	SEC	T_5	MIN	SEC	T_5	MIN	SEC	T_5	MIN	SEC	T_5
	0	1380												
	15	1330	1	15	1080	2	15	920	3	15	850	4	15	820
	30	1270	1	30	1030	2	30	890	3	30	840	4	30	820
	45	1230	1	45	990	2	45	870	3	45	835	4	45	820
1	0	1180	2	0	950	3	0	860	4	0	830	5	0	820

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TABLE II

COMPARISON OF RESULTS OBTAINED BY SEVERAL INVESTIGATIONS OF AIR-COOLED TURBINE BLADES

Investigator	Ref.	Configuration	Gas Temp.	Blade Temp. Reduction or Hot Gases Permissible Temp. Increase	Cooling Air, % Total Wt
NACA	Ref. (a)	Hollow Blade	Not Specified	Permissible Temp. Increase 580° F.	16.0%
NACA	Ref. (a)	Blade with Insert	Not Specified	Permissible Temp. Increase 790° F.	5.5%
Kohlmann	Ref. (b)	Hollow Blade	1592° F.	Blade Temp. Reduction 288° F.	10.0%
Mildahn	Ref. (c)	Cooling Jets (Boundary Layer)	1500° F.	(Limited area) 380° F.	.53 lb/min
Ness	Ref. (d)	Film Cooling, Holes and Slot in Leading Edge	1500° F.	140° F. to 285° F.	.9 lb/min
Dressendorfer	Ref. (e)	Ceramic Sleeve Air cooled	1600° F.	875° F.	.9 lb/min
Jennings	Present Report	Grooved Blade, Metal Sleeve	1460° F.	640° F. to 900° F.	1.67%

TABLE III

PRESSURE MEASUREMENTS IN TEST SECTION FOR CONFIRMATION OF MACH NUMBER

T_{11} °F.	P_{in} Hg	Fuel Flow	T_o °F.	P_s in Hg	P_o in Hg	Barom. in Hg	P_s/P_o
80	15.6	0	142.0	0.0	1.3	29.07	.961
80	15.6	86	800	0.0	2.9	29.07	.911
80	15.6	112	1000	0.0	3.4	29.07	.896
80	15.6	138	1200	0.0	3.9	29.07	.883
80	15.6	168	1420	0.0	4.5	29.07	.866

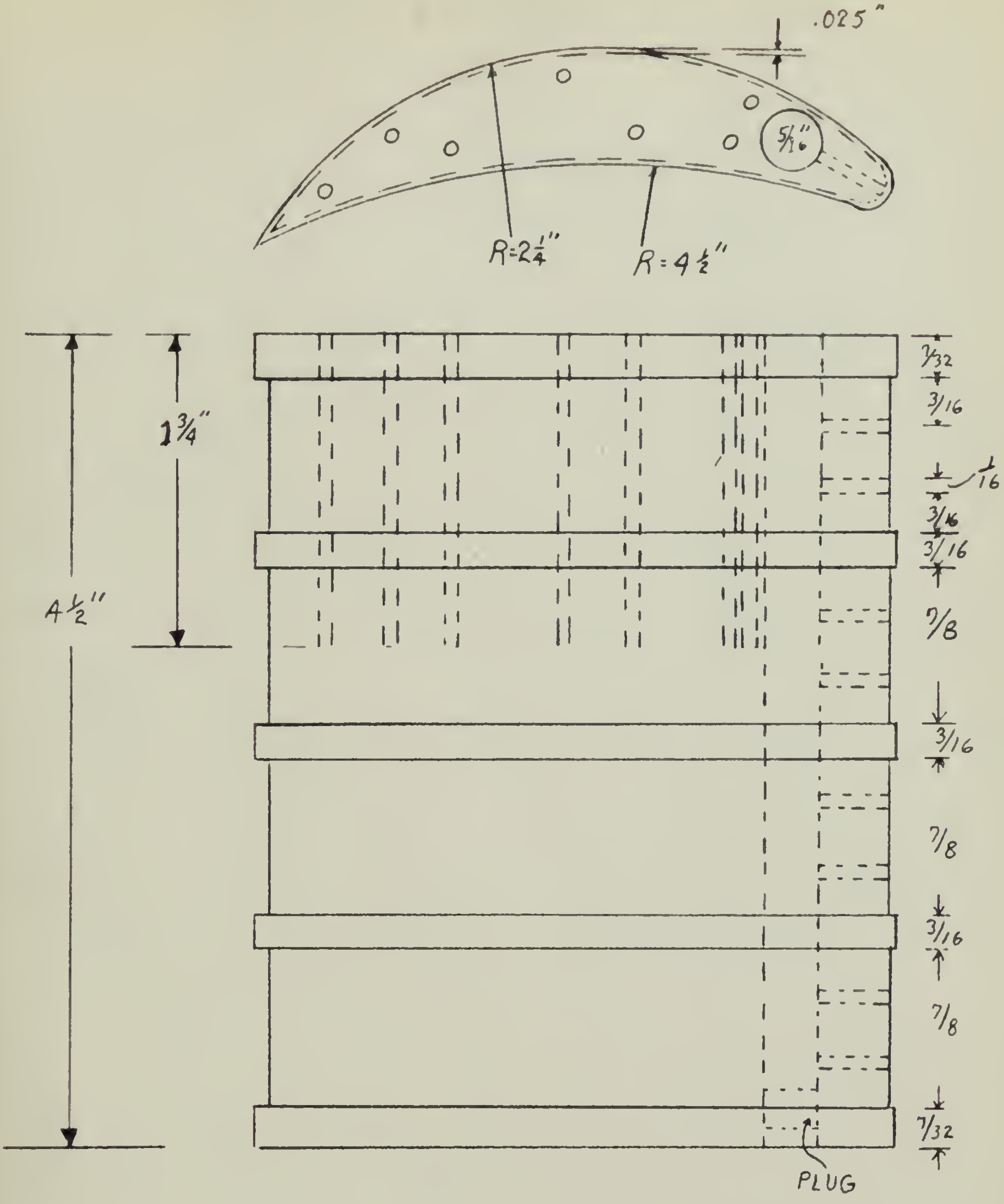
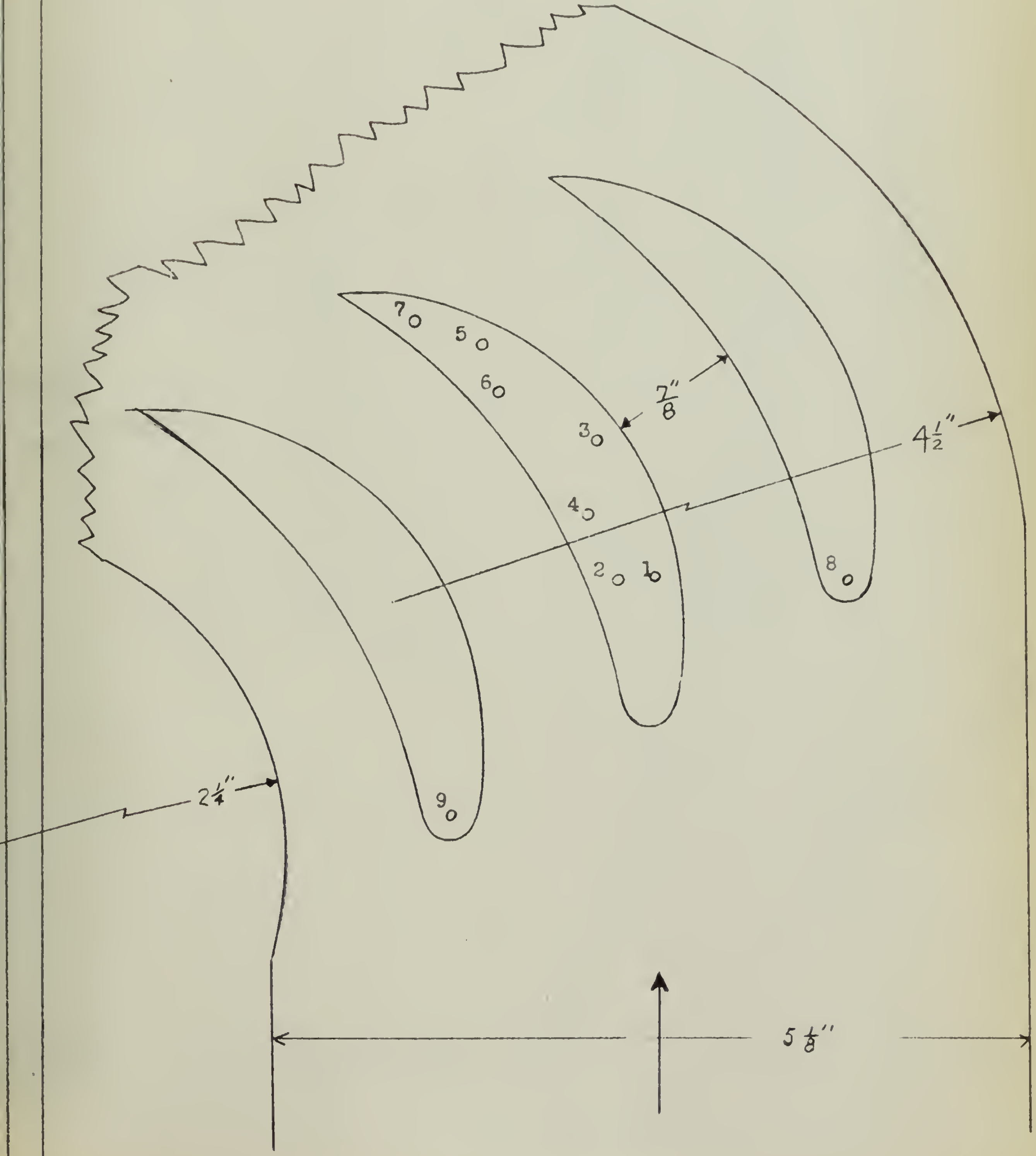


Fig. 1

Sketch of Body of Grooved Turbine Blade Model

Fig. 2

Line Sketch of Test Section Showing Static Test Cascade and Location of Thermocouples



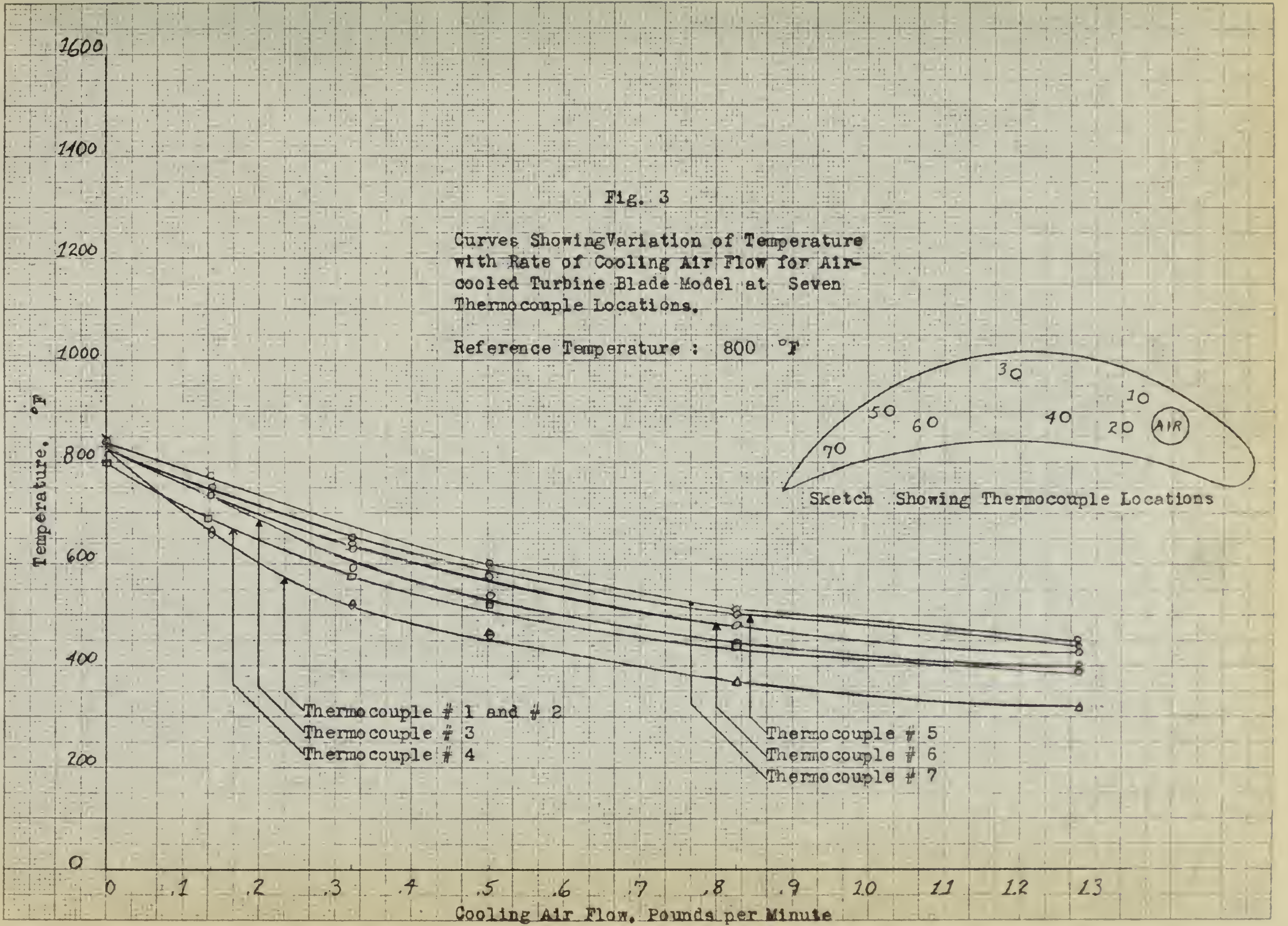


Fig. 3

Curves Showing Variation of Temperature with Rate of Cooling Air Flow for Air-cooled Turbine Blade Model at Seven Thermocouple Locations.

Reference Temperature : 800 °F

Sketch Showing Thermocouple Locations

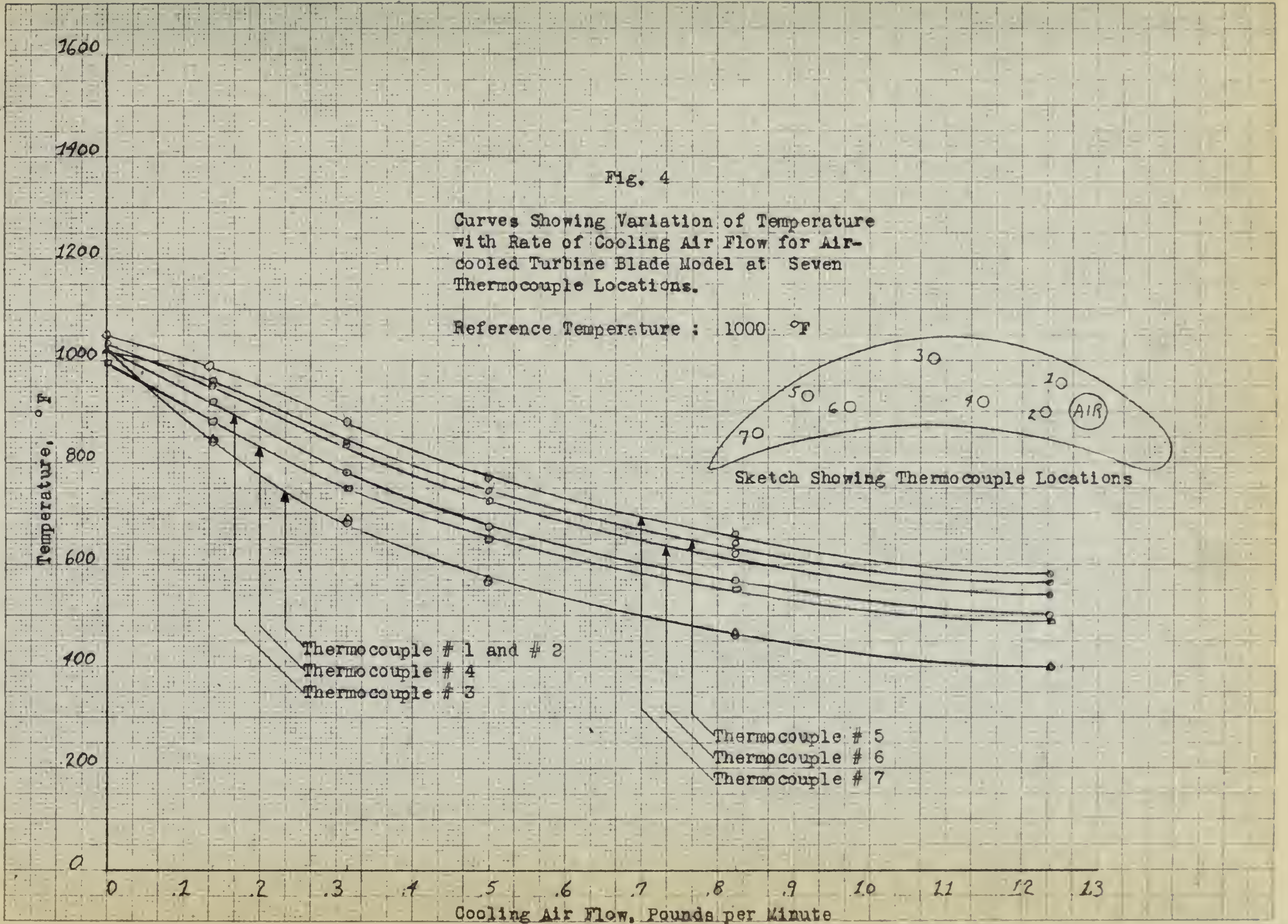


Fig. 4

Curves Showing Variation of Temperature with Rate of Cooling Air Flow for Air-cooled Turbine Blade Model at Seven Thermocouple Locations.

Reference Temperature : 1000 °F

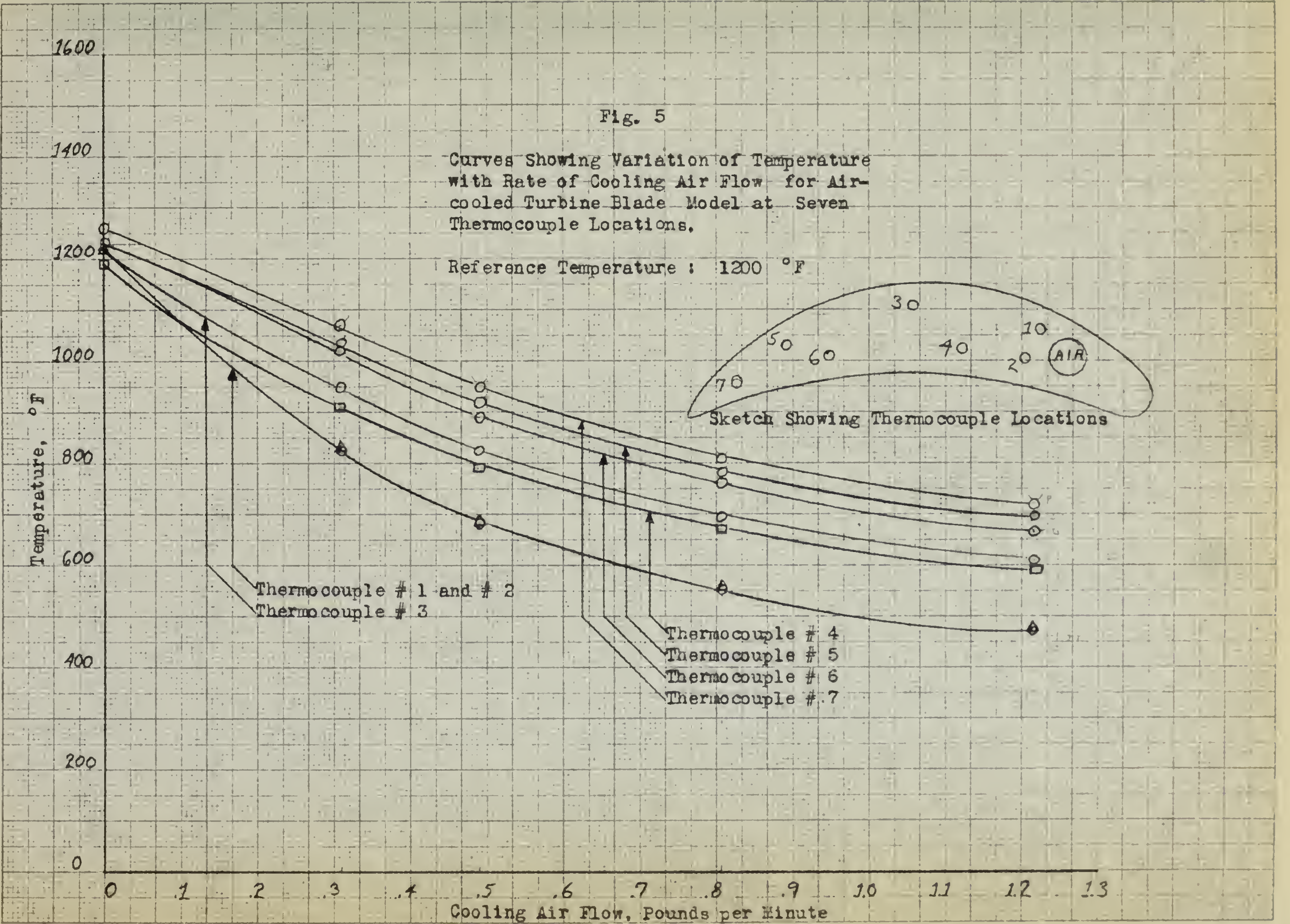


Fig. 5

Curves Showing Variation of Temperature with Rate of Cooling Air Flow for Air-cooled Turbine Blade Model at Seven Thermocouple Locations.

Reference Temperature : 1200 °F

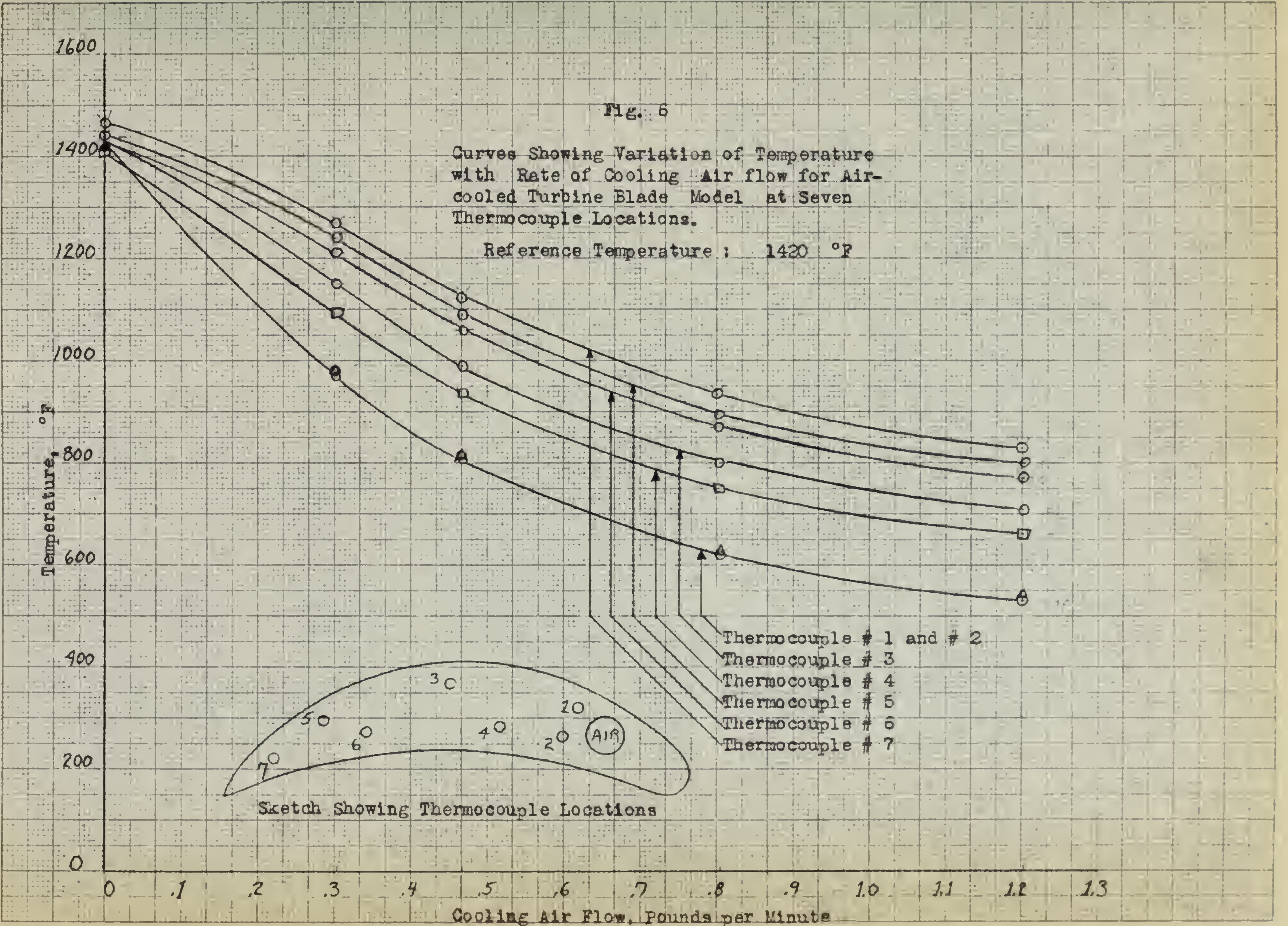


Fig. 6

Curves Showing Variation of Temperature with Rate of Cooling Air flow for Air-cooled Turbine Blade Model at Seven Thermocouple Locations.

Reference Temperature : 1420 °F

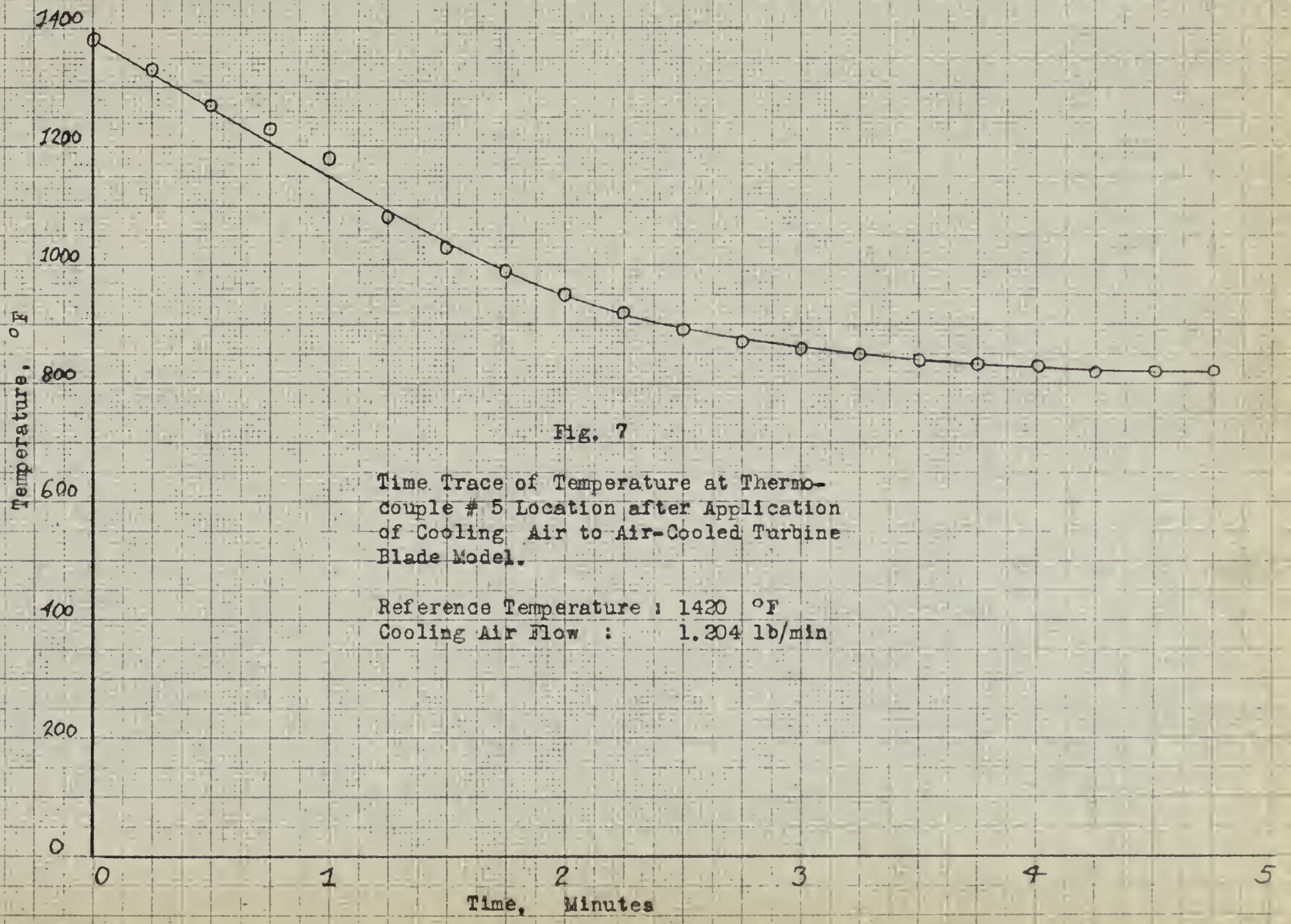


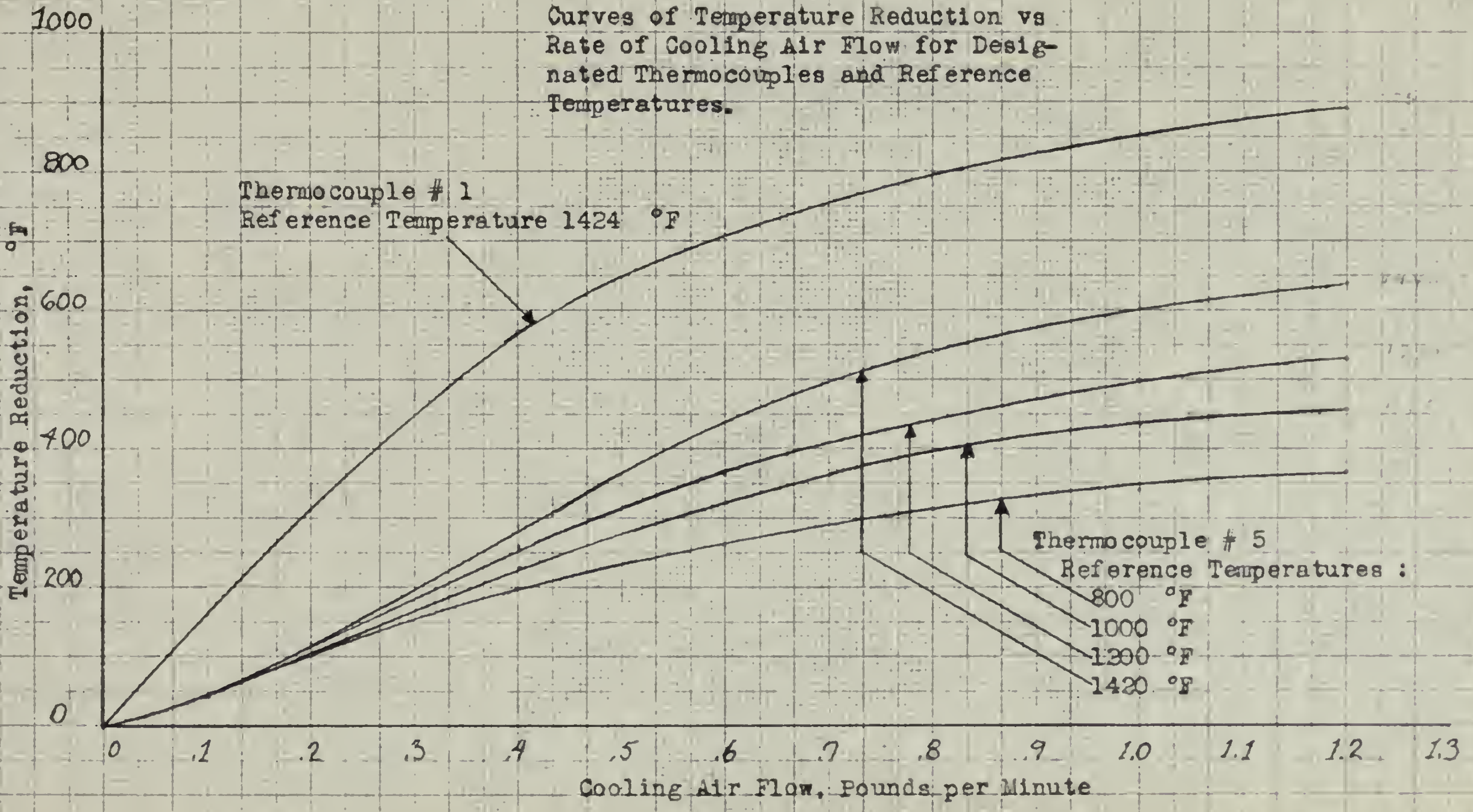
Fig. 7

Time Trace of Temperature at Thermocouple # 5 Location after Application of Cooling Air to Air-Cooled Turbine Blade Model.

Reference Temperature : 1420 °F
Cooling Air Flow : 1.204 lb/min

Fig. 8

Curves of Temperature Reduction vs Rate of Cooling Air Flow for Designated Thermocouples and Reference Temperatures.



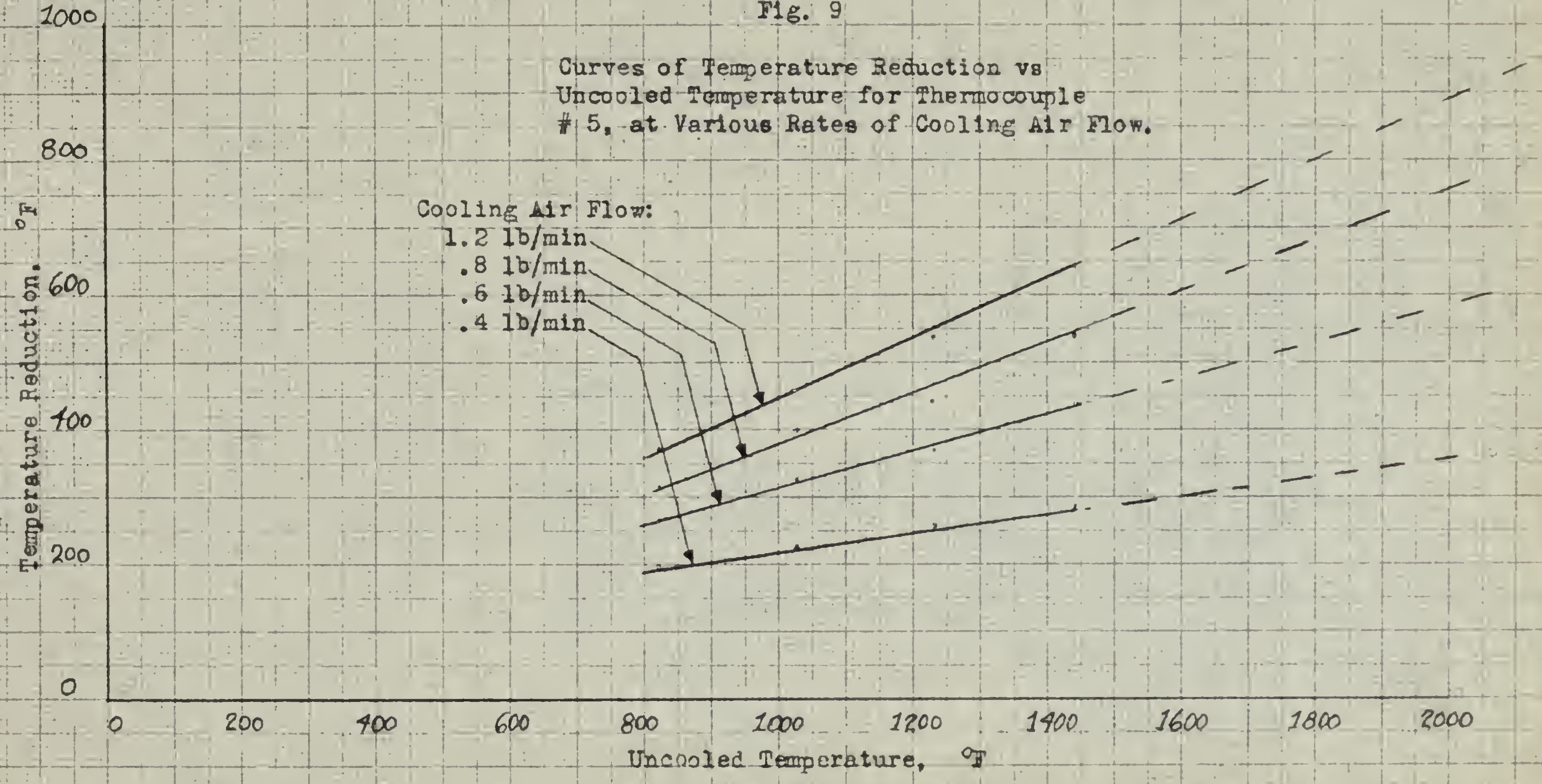
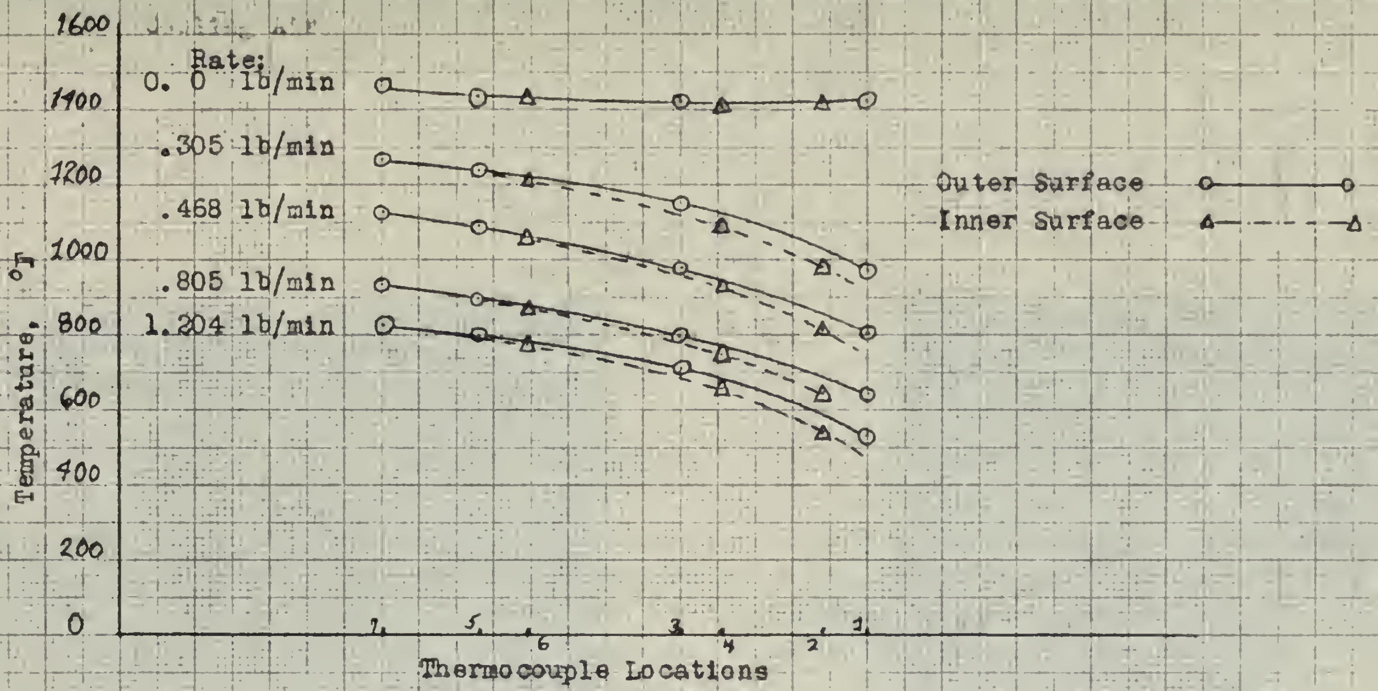


Fig. 9

Curves of Temperature Reduction vs Uncooled Temperature for Thermocouple # 5, at Various Rates of Cooling Air Flow.



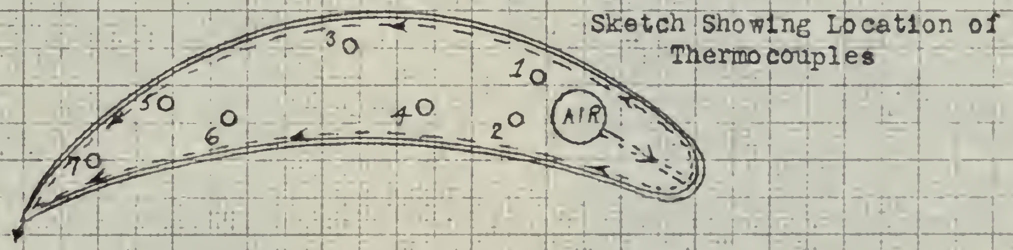


Fig. 10

Curves Showing Chordwise Temperature Distribution for Outer and Inner Surfaces of Air-cooled Turbine Blade Model with Various Rates of Cooling Air Flow.

Reference Temperature: 1420 °F

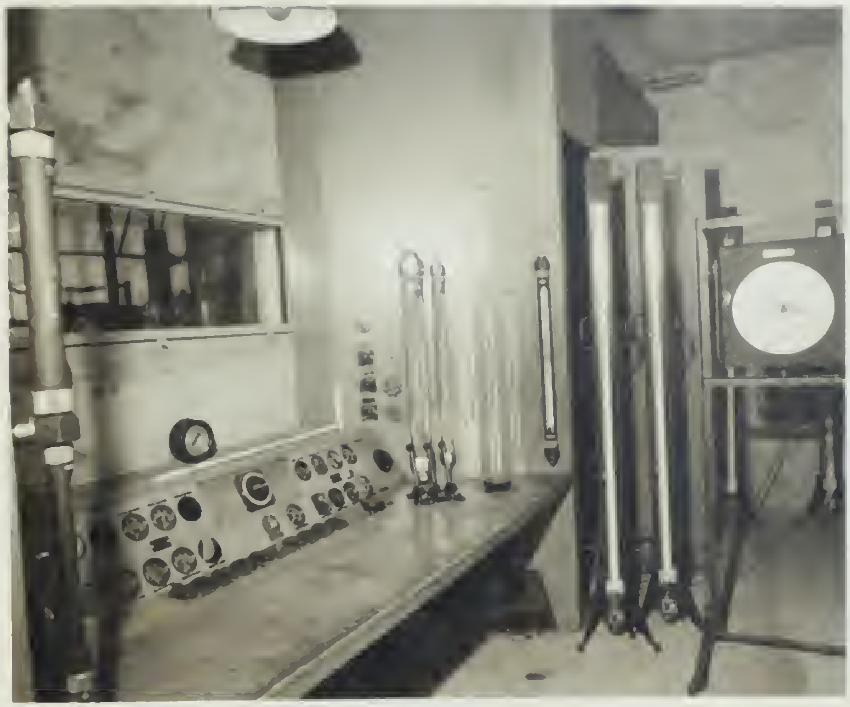


FIG. 12
CONTROL PANEL

FIG. 11
TEST CELL

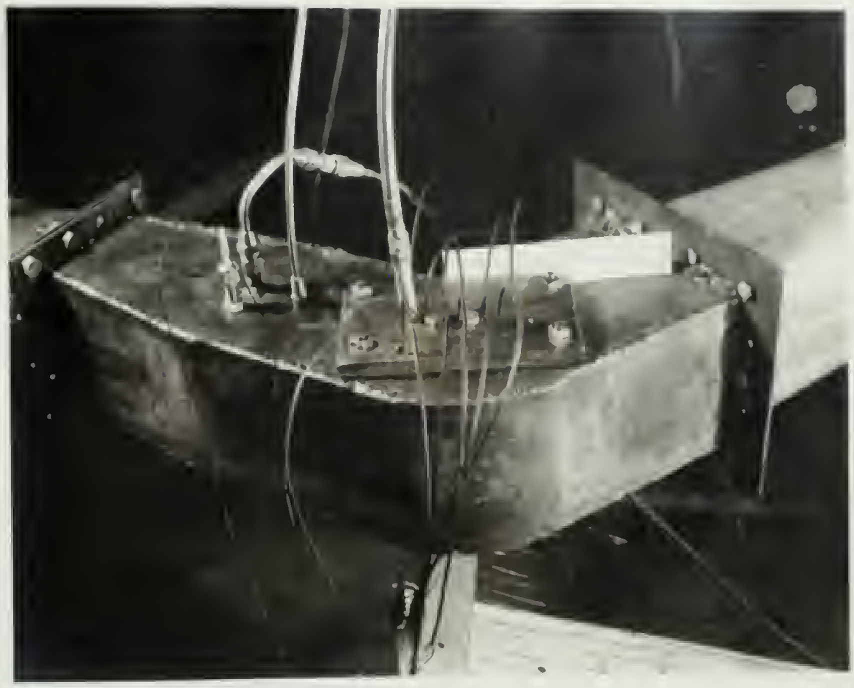


Fig 13
TEST SECTION

Fig. 14
HORIZONTALLY GROOVED AIR COOLED TURBINE BLADE MODEL WITH COVERING METAL SLEEVE

APPENDIX

The Mach Number in the Test Section ahead of the blades and in the flow about the blades was desired for reference purposes. Measurements of P_s and P_o in the test section were expected to give this information through the P_s/P_o ratio in the gas tables, interpolated for a gamma of 1.33 of the combustion gases.

The mass flow (neglecting weight of fuel) determined from the inlet orifice should also provide a check on Mach number in the test section by application of $w = \rho AV$, where ρ and A were values at the test section.

Comparison of Mach Numbers determined by the two methods did not show agreement, so the run of Table III was made to check pressure values. The Machs as determined from this second table still did not agree with the Machs as determined from the mass flow for the runs. Cause of disagreement was sought.

All pressure leads had been thoroughly checked for leaks before attachment to the test section. It is noted that total pressure agrees with measurements taken in the

first set of runs. Static pressure agreed -- but this agreement was at zero reading. It is considered that static pressure should have increased somewhat as fuel flow increased -- it was therefore decided that the P_s reading was in error, and that a leak must have occurred at the point of attachment. No pressure check for leaks was made at this point because of its position within the test section.

Further consideration showed that in view of the apparent dependability of the total pressure readings the P_s could be determined by simultaneous solution of the mass flow relations and the pressure ratio relations for the Mach Number in the Test Section. This solution was performed graphically, and the results given below:

T_0	Mach Number at Test Section	Mach Number Around Blades
800° F.	.265	.400
1000° F.	.284	.435
1200° F.	.299	.460
1420° F.	.314	.49

Mach Number around the blades was determined from the area relation of the test section cross section (23 sq. in.) to the area presented for flow in the cascade (15.75 sq. in.).

TABLE OF REFERENCES

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Ref. (c) "Air Film Cooling of a Metal Surface Exposed to High Temperature and High Velocities", by E. C. Mildahn, M.S. Thesis, University of Minnesota, August 1950.

Ref. (d) "Boundary Layer Control as a Method of Gas Turbine Blade Cooling", by D. O. Ness, M.S. Thesis, University of Minnesota, August 1949.

Ref. (e) "A Study of the Heat Transfer Characteristics of a Turbine Blade Having a Ceramic Sleeve with Air Cooling", by D. E. Dressendorfer, M.S. Thesis, University of Minnesota, August 1949.

Ref. (f) "Introduction to Heat Transfer", by A. I. Brown and S. M. Marco, 1st edition 1942, McGraw-Hill, New York.

Ref. (g) "Flow Measurement 1940", American Society of Mechanical Engineers, New York.

SAMPLE CALCULATIONS:

1. Cooling air flow rate:

$$Q_2 = Q_1 \left(\frac{p_2}{p_1}\right)^{\frac{1}{2}} \left(\frac{t_1}{t_2}\right)^{\frac{1}{2}}$$

"FLOWRATOR" equation from instrument handbook.

For the first run at 800°F:

$Q_2 = 8.05$ (METER READING)
$p_2 = 14.7$ psia (METER CALIBRATION)
$p_1 = (60 + 14.32) = 74.32$ psia
$t_2 = 560°R$ (METER CALIBRATION)
$t_1 = (75 + 460) = 555°R$ (COOLING AIR)

$$Q_1 = 8.05 \left(\frac{74.32}{14.7}\right)^{\frac{1}{2}} \left(\frac{560}{555}\right)^{\frac{1}{2}} = 18.0 \text{ ft}^3/\text{min}$$

@ 100°F, 14.7 psia

$$w = \rho Q = (.071)(18) = \underline{1.278} \text{ lb/min}$$

2. BURNER AIR FLOW: Ref. (g)

$w = .668\, A_2 K \sqrt{\rho_1 \Delta P}$, which for the orifice used reduces to

$$w = 2.52 \sqrt{\frac{P\, \Delta h_w}{T}} \text{ lb/sec}$$

P = Barometer, inches of mercury
T = Room intake temperature, °R
Δh_w = Orifice pressure drop, inches of water
w = Air flow rate, lb/sec

$$w = 2.52 \sqrt{\frac{29\ 15 \times 15.6}{565}} \text{ lb/sec}$$

$$w = \underline{2.26} \text{ lb/sec}$$

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